

What are Orbit plots, anyway?

by Mark A. Jordan
MDS Vibration Specialist
Bently Nevada Corporation

One question often asked by plant personnel is... "Just what are Orbit plots, anyway, and how can they help me solve machinery problems?"

There are many ways to observe signals generated by noncontacting proximity probes, including Bode and Polar plot formats. These plots establish a rotor's frequency filtered amplitude and phase components, through transient and steady state operations. However, an understanding of Orbit and Average Shaft Centerline Position plots helps you indicate how the dynamics of machinery malfunctions takes place, and how they can be more accurately identified before failure.

Therefore, monitoring Orbit and average shaft centerline position within a bearing provides important and relevant information on rapidly changing machinery conditions.

Many vibration transducers are available in today's marketplace. Choosing the correct transducer for a specific application is not only crucial for accurate machinery vibration monitoring but also for diagnostic capabilities.

Bearing cap vibration information cannot truly indicate the dynamic response of the shaft in a state of malfunction. Casing measurements acquired by seismic transducers (either velocity or accelerometer) can be grossly inaccurate.

Therefore, using a case-mounted transducer system by itself can only be viewed as an indirect method of quantifying a machine's malfunction.

Conversely, proximity probes can measure the *direct* relative response of the rotor to the stationary bearing housing. And for those machines that possess high bearing cap activity, both a proximity probe *and* a casing transducer can be used, resulting in what is known as *shaft absolute* motion. The term "absolute motion" has historically been used because antique shaft riders originally yielded this reading. Unfortunately, shaft riders are unreliable and can't show slow roll data. This severely limits their use in machinery diagnostics, and even more so in balancing.

When noncontacting eddy current probes and Proximitors are used to monitor lateral shaft motion, the proximity probe provides the following signal components:

- A dc signal proportional to the average shaft position relative to the probe mounting.
- An AC signal (in this case, negatively fluctuating) corresponding to shaft dynamic motion relative to the probe mounting.

In typical plant applications, transducer signals are usually processed and displayed by a radial vibration monitor. Proximity probes are primarily used on machines with fluid film-lubricated bearings, such as turbines, pumps and compressors.

For minimum machinery monitoring, two orthogonally-mounted proximity probes should be mounted at each bearing. This provides the required AC and dc signals for on-line monitoring and diagnostics. When used in conjunction with a once-per-turn reference probe

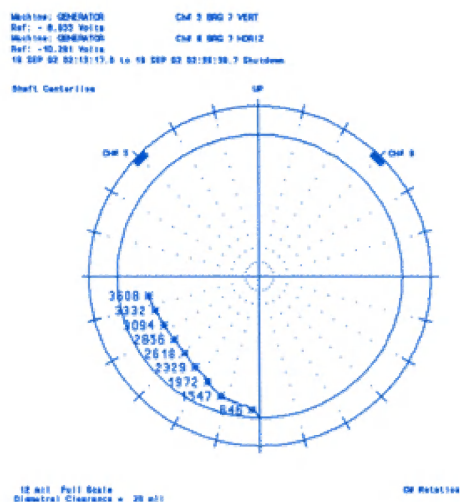


Figure 1

Shows the Shaft Average Centerline Position within the bearing clearance using two orthogonally-mounted displacement probes. For clockwise rotation, this response is considered acceptable.

(Keyphasor[®]), the diagnostic capability is even more pronounced. These transducers provide most of the machinery data needed for proper rotating machinery monitoring, such as Orbit and Average Shaft Position. However, mode identification probes, installed at each end of a machine or at each radial bearing, should be considered for more complete information and problem diagnosis. More information on mode identification probes is available in Bently Nevada Applications Note AN040 or in the following issues of the *Orbit*: "Shaft observing mode identification probes for improved machinery protection," Sept. 1990, p. 10. and "Mode identification probes," Feb. 1991, p. 31.

dc component - Average Shaft Position

Average Shaft Position is the average position of the shaft relative to the stationary component on the machine where the probe is mounted. Voltage fluctuations are generated by the proximity probe relative to the distance change caused by dynamic rotor motion under operating conditions. To obtain accurate shaft centerline data you must reference voltage changes to a zero machine-speed gap voltage reference. For horizontal machines, this reference is generally obtained with the rotor at rest or on turning gear.

In this condition, the rotor is assumed to be at rest in the bottom of its bearings; therefore, all subsequent gap voltage changes are referenced to this starting position.

As machine speed increases during startup, changing gap voltages from two orthogonally-mounted probes indicate the amount of average shaft travel within that bearing clearance. At running speed, the rotor's average position within the bearing is easily identified when the referenced zero speed gap values are used. By analyzing Average Shaft Position within the known diametral bearing clearance, valuable information regarding alignment, overall bearing condition, oil film thickness, shaft radial loading, etc. becomes available.

AC component - Orbits

The AC component of the transducer signal produces a periodic waveform, one from each of the two orthogonal probes. A typical output waveform is shown in Figure 2.

Note that in Figure 2, two separate waveforms are shown; the left waveform, filtered to running speed (1X), shows a smooth sine wave, while the right represents unfiltered overall vibration. In establishing how Orbits are formed, you must first know that the waveform produced by each transducer is an individually-processed vibration signal. This signal is generated at a spe-

cific angular location on the rotor which relates the amount of shaft lateral motion in that plane. When two probes are mounted orthogonally (XY configuration, 90° apart), the two individual signals (waveforms) are representative of shaft peak-to-peak displacement in their respective angular planes and are plotted as amplitude, or displacement, versus time. (Figures 2 and 3).

An Orbit is generated by pairing together the two XY waveforms so the time element is removed, leaving the X amplitude component versus Y amplitude component, plotted in what is commonly known as the Cartesian Coordinate (or polar coordinate) sys-

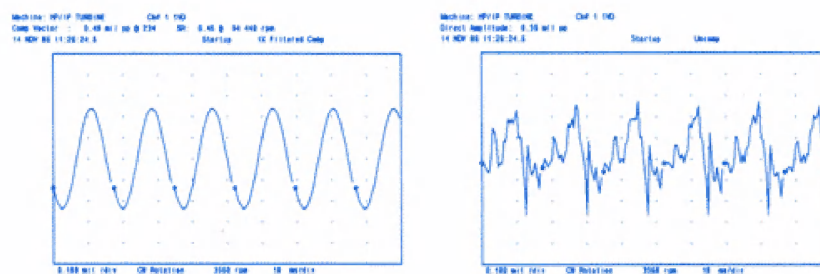


Figure 2

The left waveform represents the rotor's synchronous (filtered to 1X) lateral vibration response, while the right is a representation of overall vibration present in the system. Frequencies in the overall timebase are from dc (0 Hz) to 10 kHz inclusive.

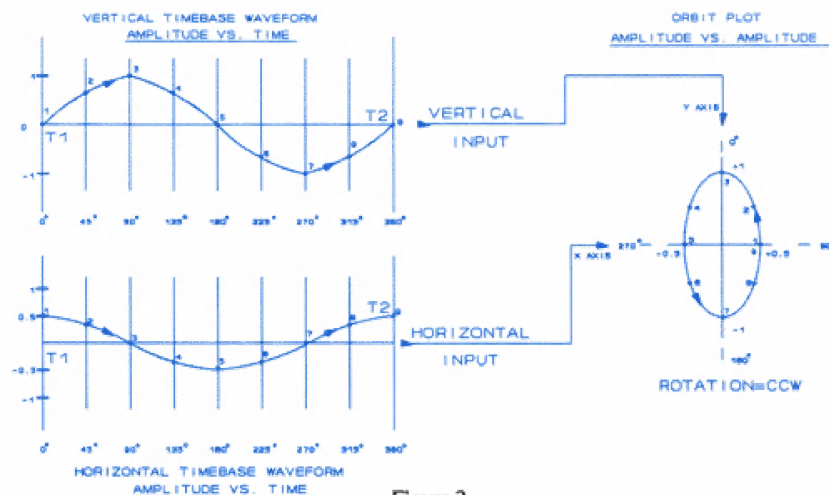


Figure 3

The graphical result of plotting Equations 1 and 2 from time T1 to time T2. To the right of the waveforms, the associated shaft orbit plotted as amplitude versus amplitude. Numerical points (1, 2, 3, etc.) along the Timebase waveforms correspond to specific points on the Orbit precession. The same inputs are used for Figure 4.

tem. To illustrate this point, take a pair of XY Timebase waveforms, which are separated by a phase difference of 90° and whose waveform amplitudes are 1.00 mil (from the vertical probe) and 0.50 mil (from the horizontal probe). These two signals are described by the following equations:

$$X(\theta)_{\text{horizontal axis}} = 0.50 \cos(\theta) \quad (\text{Equation 1})$$

$$Y(\theta)_{\text{vertical axis}} = 1.00 \sin(\theta) \quad (\text{Equation 2})$$

Where $\theta = \omega t$ (ω = rotational rotor frequency, t = time) represents one shaft revolution (T1 to T2) in radians

and the numerical values (1.00 mil & 0.500 mil) are the amplitudes of lateral shaft vibration.

In Figure 3, Equations 1 and 2 are plotted in the amplitude versus time domain (waveform). Similar results are achieved using a machine's 1X filtered XY waveform pair or unfiltered waveform pair for each bearing, which yields a 1X filtered Orbit or an overall vibration Orbit, respectively.

Waveforms and orbital presentations can be easily displayed on a two channel oscilloscope. It is important to note that the oscilloscope should have a third channel, a "Z" channel for a Keyphasor

signal input (Figure 4). When two vibration signals are input to a dual channel oscilloscope and observed on its display, the amount of vibration can be displayed in timebase (sinusoid waveform) or in orbital form (Figure 4).

In its Orbit mode, the oscilloscope places the vertical (Y) and horizontal (X) signals along their respective axis to create a display of amplitude versus amplitude. The form in which this takes place is governed by the following equations:

$$X(r, \theta)_{\text{horizontal axis}} = r \cos(\theta) \quad (\text{Equation 3})$$

$$Y(r, \theta)_{\text{vertical axis}} = r \sin(\theta) \quad (\text{Equation 4})$$

Where $\theta = \omega t$ (ω = rotational rotor frequency, t = time) represents one shaft revolution (in radians), and r denotes lateral shaft amplitude.

An Orbit pattern, as seen on an oscilloscope, is simply a light beam dot moving very rapidly so it looks like a continuous line on the screen. This rapidly moving dot represents the centerline motion of the shaft as seen by the proximity probes. The Orbit is the path of the rotor centerline at the lateral position of the probes.

The Keyphasor® pulse, when fed to the "Z" intensity input of the oscilloscope, intensifies the dot at the instant in time when the keyway (once-per-turn event) is passing under the Keyphasor probe. Therefore, the Keyphasor® dot on the Orbit (or waveform) represents the centerline location of the shaft in its path of travel (or high spot) at the instant that the keyway is in front of the Keyphasor® probe.

This technique identifies a fixed physical reference to the shaft. This arrangement produces not only peak-to-peak amplitude, but important phase information that is commonly used in machinery diagnostics. Figure 5 shows actual machinery field data processed by a vibration diagnostics software package. The Average Shaft Position within the bearing clearance and the Orbit's elliptical shape indicate rotor loading and/or differences in dynamic stiffness at a bearing location.

Notice the Orbit plot in Figure 5 is slightly elliptical. This data suggests the

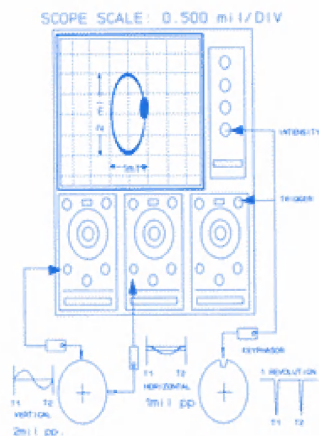


Figure 4

Two individual timebase signals input into an oscilloscope along with a Keyphasor® signal, allow angular phase information to be represented as an Orbit. The Orbit amplitude spans 2 mil peak-to-peak vertically (Y direction) and 1 mil horizontally (X direction) with the XY channel amplitude scales set at .5 mil per division.

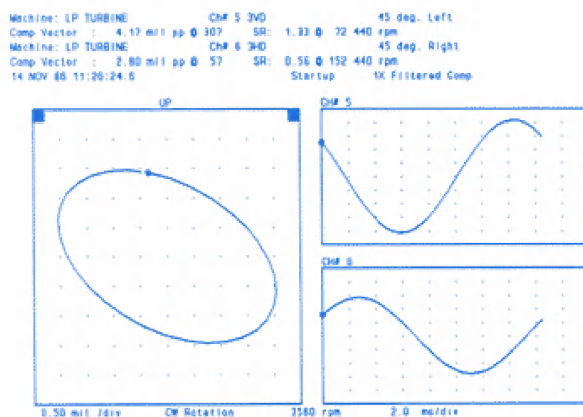


Figure 5

Shows one revolution of the rotor with the Timebase waveforms to the right and the associated Orbit to the left.

rotor is in good operational condition with normal minor influential forces, such as gravity, fluidic, and bearing load forces.

Shaft position and Orbits

Machinery diagnostics depends on knowledge of a machine's bearing parameters. Elements, such as diametral clearance and specific bearing type, are helpful when applying the diagnostic techniques discussed here.

Cases 1, 2 and 3 illustrate changing machinery conditions. Each case shows the relative shaft position within the bearing clearance, along with its associated Orbit. These examples progress from normal operation to a state of malfunction and are taken from actual machine data. They have been redrawn for clarity.

The Orbit/Shaft Centerline data shown in Figure 6 (Case 1) shows a machine in good operating condition. The Shaft Centerline data shows the rotor is in the lower left quadrant. The Orbit does not show evidence of abnormal loading.

The next set of data, however, (Figure 7) demonstrates how a malfunction, such as misalignment between two machines, can affect both the shaft position and Orbit display as a result of shaft preloading. The change of shape of the Orbit can, for example, indicate changing preloads (i.e. misalignment) acting on the rotor. If the restraining forces (the dynamic stiffness) in the machine are equal in all radial directions, with the only force acting on the rotor its residual imbalance, then the Orbit should be completely circular. Other forces or unequal restraints cause the rotor to respond in noncircular shapes, such as those illustrated in Figures 7 and 8.

A progressive preload, (Figure 8), may result in the Orbit shape changing into a "Figure 8." The associated shaft average position is shown in the upper left quadrant of the Shaft Centerline plot. The Figure 8 shape, and the fact that the average shaft position is now in the upper left quadrant, indicate excessive rotor loading that may be a result of misalignment coupled with excessive pipe strain

and/or severe bearing problems. A machine train experiencing such problems should be inspected immediately.

Preloads - a general summary

These three cases show how shaft centerline position and Orbit shape might respond to an increasing steady state unidirectional preload, progressing from a normal condition to a heavy preloading, resulting in the classical Figure 8 pattern. By observing orbital patterns over time, the degree and plane of a preload condition can be determined and tracked.

Typically, a heavy preload is not indicated by a perfectly shaped Figure 8, but by different sized loops. Preloads affecting a rotor system can fall into several different categories. Radial preloads include gravity, fluidic forces, abnormal bearing loads (especially internally adjustable types), seals, and pipe strain.

Preloads can be damaging if the forces involved are strong enough to create fatigue. The results can be seen in excessive bearing wear, shaft fatigue, shaft cracks, etc.

In observing Orbit shapes, increasing rotor preloads will force the Orbit into ➤

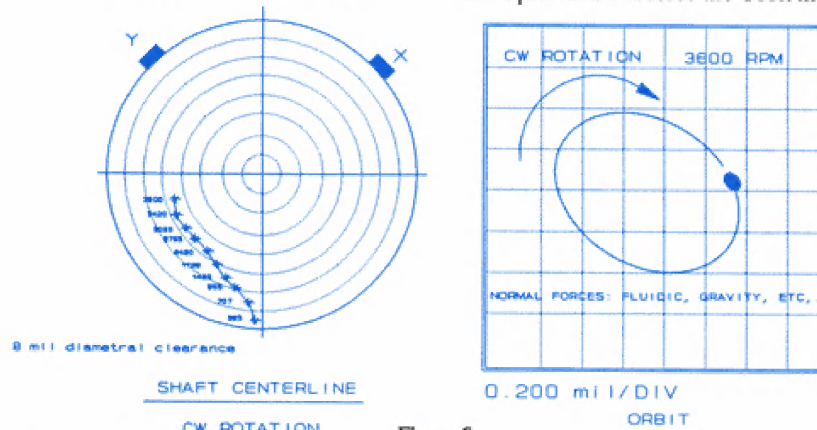


Figure 6

An acceptable unfiltered Orbit at 3600 rpm along with its associated shaft centerline travel in the bearing during startup. Notice the machine rotation is clockwise, as indicated by the shaft position in the lower left quadrant of the Shaft Centerline plot. (for CCW rotation, this shaft position would be in the lower right quadrant). This Orbit/Shaft Centerline plot is representative of a horizontally-mounted machine in a good operational state.

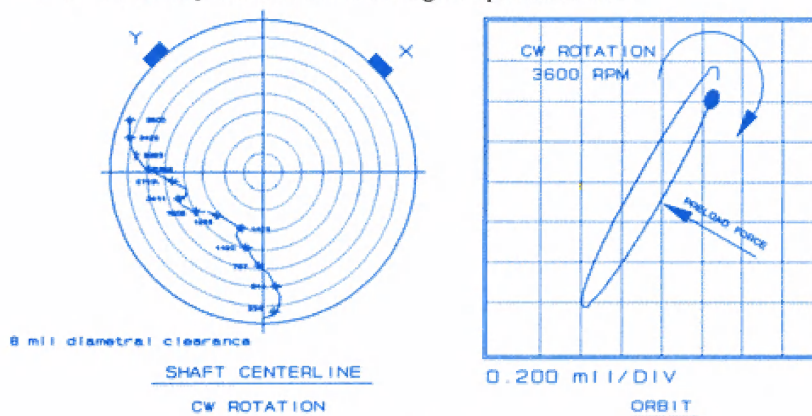


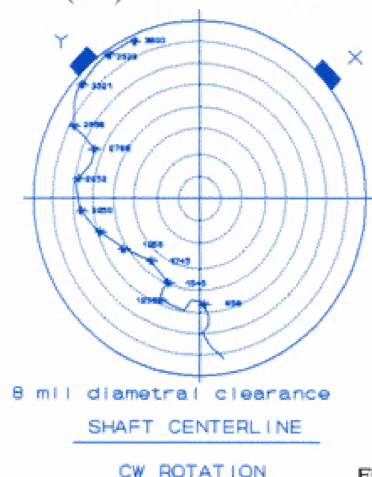
Figure 7

A condition of lateral loading malfunction evidenced by shaft centerline travel into the upper left quadrant, and an elliptical Orbit shape caused by a heavy resultant force acting on the rotor. Conditions such as this are typically seen on the inboard bearings of turbines/compressors when misalignment is present. Additionally, this may also be seen on the No. 1 Bearing of steam turbines, mainly due to the summation of dynamic forces present i.e. steam flow, gravity, bearing preload, fluid properties, etc.

a more elliptical shape. The initial response is a change in the 1X amplitude, followed by an increase in other frequencies, such as 2X, indicating severe misalignment or other malfunctions are present.

Orbit direction & vibration precession

Figure 9 shows two Orbits of the same size and shape but with different orientations and precessions. Orbit "A" might be considered normal for a bearing with the journal rotating counterclockwise (CCW) but abnormal for clockwise rotation (CW). The converse is true for



Orbit "B" — fairly normal for CW rotation, and abnormal for CCW rotation.

Note the Keyphasor® dot and the preceding "blank" section in the Orbit precession. From this mark, the direction of vibration precession can be determined. The term blank-bright implies the vibration direction given by the Keyphasor® location on the Orbit precession. The machine's actual rotational direction under normal conditions should coincide with vibration direction.

Common malfunctions

Orbit data presentations are of great

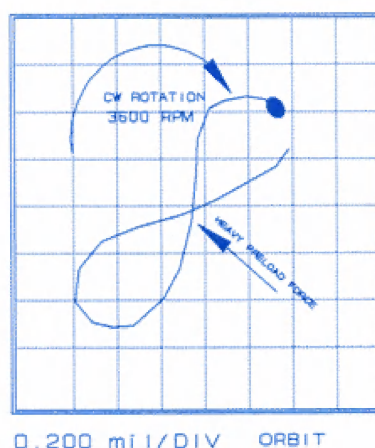


Figure 8

Shows a condition of heavy loading evidenced by shaft centerline travel into the upper left quadrant and against the bearing wall. A Figure 8 Orbit shape suggests reverse precession, indicating the preload condition is severe.

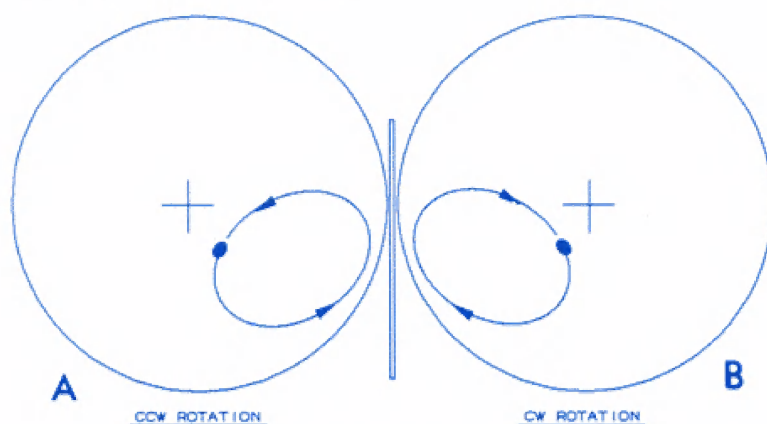


Figure 9

Orbit precession (shown inside the bearing clearance). The Keyphasor® blank-bright spot shows the direction of vibration precession when viewed on an oscilloscope. (Note, this serves to illustrate the rotor direction of two different machine trains).

value to the machinery vibration specialist. Many different types of malfunctions can be identified through Orbit analysis. A few of these malfunctions are illustrated here to show how much can be learned by using this powerful diagnostic tool.

Figure 10 illustrates a rotor with minimal preloads (other than normal loading forces, gravity, fluidic, etc.), whose predominant frequency is 1 times running speed.

Lateral load malfunctions

When misalignment (the most common preload cause) is present, the Orbit might resemble that shown in Figure 11. Increased angular and/or parallel offsets between two rotors are the most common causes of machinery misalignment.

Fluid induced instabilities (whirl & whip)

Figure 12 shows a subsynchronous component, a fluid instability, as illustrated by the two Keyphasor® dots. The precession of the vibration component is always forward (same direction as machine rotation), and the Orbit shape is usually circular.

For a typical fluid film bearing, the self excited vibration frequency of an instability is typically within the range of 30% to 48% of machine operating speed.

When fluid whirl is viewed on an oscilloscope, the two Keyphasor® dots rotate slowly against machine rotation, indicating that the subsynchronous frequency is less than 50% of machine speed. Conversely, if the dots remain stationary, then the frequency is exactly half (50%) of machine speed.

Figure 13 shows a destructive bearing malfunction in progress. Typically known as fluid whip, its vibration precession is forward. The period of this self-excited malfunction may or may not be harmonically related to the rotational period of the shaft. When it is not, the Keyphasor® dots move in a seemingly random pattern; when it is harmonically related, the multiple Keyphasor dots will appear to be stationary. This condition is accompanied by large

vibration amplitudes that usually traverse the bearing clearance.

Rotor contact malfunctions - rub

Shaft position and Orbit shapes can also indicate rubs, another form of machinery malfunction commonly seen in today's applications. Rubs occur when the rotating shaft contacts a stationary part of the machine. Malfunctions include shaft contact with seals (usually with minimal radial clearances), newly-installed steam packing, contact of turbine/compressor blading due to a failed thrust bearing, abnormal case anomalies due to thermal warping, etc.

A rub occurs as a secondary effect of a machinery malfunction. It is indicated by increased vibration levels and a change of Orbit shape and average shaft centerline position. If the rub continues, this malfunction response becomes dominant. Unlike fluid-induced whirl and whip, a rub can have many different Orbit shapes. These shapes can range from looping Figure 8s, with increasing amplitudes over time, to a completely circular Orbit shape (full annular rub which fully encompasses the seal or bearing clearance that the rotor is in contact with).

Increased system temperature can also indicate a machine is malfunctioning. Lubrication oil and process temperatures may rise because of heat caused by friction in the system. Vibration levels may increase throughout the machine train, most likely due to shaft bow.

The two main types of rub malfunction may be classified as follows:

Partial rub. This is the most common form of rub, which occurs when the rotor occasionally contacts a stationary part of the machine. The fundamental frequency is most often at 50% of running speed. Therefore, the Orbit may resemble a Figure 8, but with two Keyphasor® dots present. Due to irregular motion of the shaft because of partial rotor contact with the stationary part(s), other frequencies may also appear in the Orbit's progression. Sub-multiples of running speed, such as .25X, .32X, etc., may be identified by additional Keyphasor® dots. For example, a 33% (1/3X) frequency may

appear as a twisted looping Orbit with three Keyphasor® dots in its vibration precession (Figure 14).

The following relationships can help identify partial rubs:

Common Frequencies Generated by Partial Rub

When	Frequencies Generated
$\Omega < 2 \omega_{\text{resonance}}$	1X
$\Omega \geq 2 \omega_{\text{resonance}}$	1X or 1/2X
$\Omega \geq 3 \omega_{\text{resonance}}$	1X, 1/2X or 1/3X
$\Omega \geq 4 \omega_{\text{resonance}}$	1X, 1/2X, 1/3X or 1/4X.

With other ratios possible

Where: Ω is rotor speed and ω is rotor first balance resonance.

Full annular rub. This rub is most often encountered in seals when the seal clearance interferes with the rotating element. In a full annular rub, the precession of vibration is backwards or opposite to the rotor direction. This condition is very destructive, and can be recognized by its Orbit shape traversing the *full extent of the bearing or seal clearance*. Once initiated, the vibration may worsen until the system restraint forces overcome the rub excitation force or the machine self-destructs. When this ►

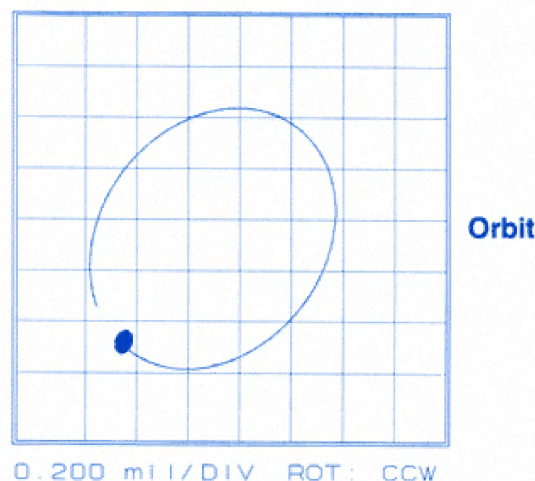


Figure 10

Representative of a machine with minimal preload characteristics and no abnormal problems. Predominant amplitude frequency is 1X.

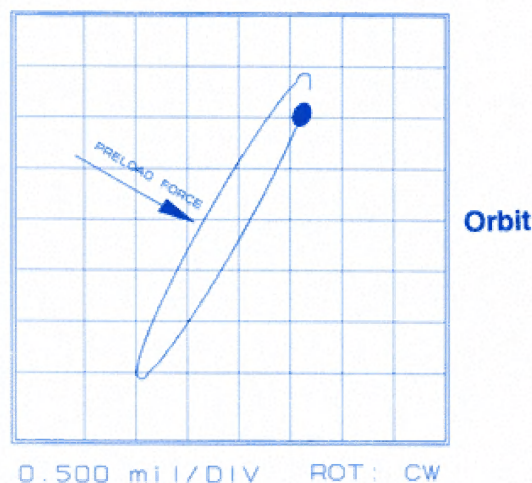


Figure 11

Representative of a machine with poor alignment, or a "cocked" bearing assembly. Further investigation warranted. Note, large amplitude present in the major plane of vibration.

malfunction is present, the machine is in imminent danger of total destruction.

When the rotor and seal are in contact, the two major regimes of the rotor vibration under steady state conditions are:

1. A forced synchronous precession due to imbalance excitations. The rotor bounces inside the seal, the lowest frequency being 1X.
2. Forced self-excited circular reverse precession with a frequency corresponding to one of the natural frequencies of the rotor system.

Therefore, the Orbit is predomi-

nantly circular to the limit of the seal clearance and has a reverse precession. The only recommended course of action under these circumstances is to shut the machine train down, inspect the damage and correct the initial problem.

A case history

The following case history shows why Shaft Centerline and Orbit plots are useful. The machine consists of an industrial gas turbine driving a six-stage, water flood injection pump. During the last few years, this pump had experienced high vibration problems. Investigation by sev-

eral independent consultants, using bearing cap vibration transducers, confirmed that the machine's vibration levels began to rise when the machine approached operating speed.

Bently Nevada's Machinery Diagnostics group was contracted to perform an in-depth vibration analysis on this machine in 1992. XY proximity transducers were temporarily installed adjacent to both pump bearings, observing exposed sections of the pump shaft.

Machinery vibration information was acquired using proximity probes. A sample Orbit is shown in Figure 15. Notice that it is a predominantly preloaded Orbit, where the major plane of vibration is oriented in a horizontal direction, suggesting that an almost true vertical misalignment condition is present.

An optical alignment study performed on this machine showed the pump was .005 inch (.13 mm) too low in relation to the turbine centerline. Furthermore, the shaft average centerline data showed the rotor was initially positioned in the upper right quadrant (Figure 17). This data indicated that the journal was originally oriented incorrectly with respect to the bearing due to system misalignment. The pump was subsequently realigned to the turbine, resulting in the change as shown in Figures 16 & 17.

Shaft Position and Orbit plots clearly show the diagnosis of misalignment was correct. An acceptable running condition was achieved through a logical diagnostic approach. The change in Shaft Average Centerline Position is clearly shown in Figure 17.

Conclusions

Although machinery needs are continuously changing, one thing remains constant: the need for a reliable method of accurately monitoring industrial machinery. Observing signals from proximity probes in their simplest form is basic to quality machinery monitoring and machinery diagnostics. Many parameters needed in complex machinery malfunction analysis cannot be determined accurately by any other means.

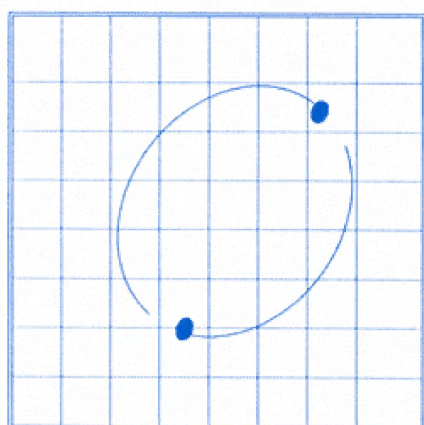


Figure 12

Representative of a machine with subsynchronous component at approximately half running speed. Usually suggests machinery self-excited fluid whirl. Note - TWO Keyphasor® dots. Usually accompanied by larger vibration amplitudes.

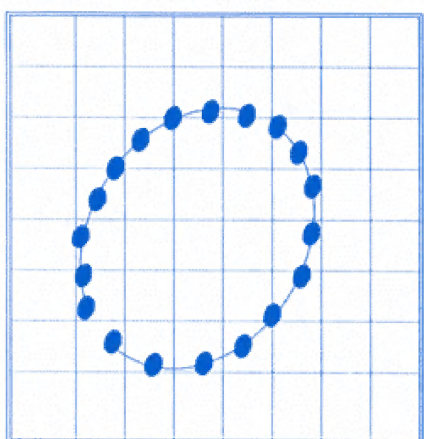


Figure 13

Representative of a machine with subsynchronous excitation at a first balance rotor resonance when the machine speed is well above two times that resonance. Typically referred to as fluid whip. Note - Multiple Keyphasor® dots - This malfunction can be very destructive, with vibration amplitudes approaching the diametral clearance of the bearing.

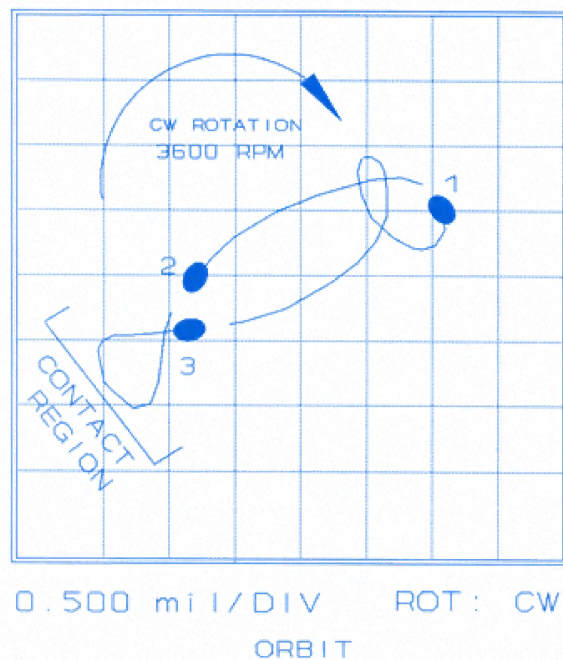


Figure 14

Representative of a machine with 1/3X subsynchronous excitation caused by a partial rub condition. Note the positioning of the Keyphasor® dots.

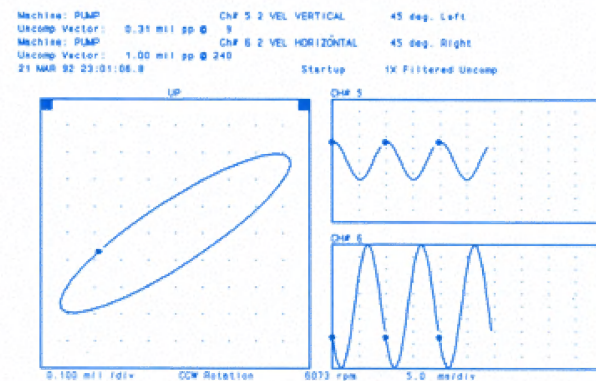


Figure 15

Shows an elliptical Orbit. Notice the preload force is predominantly oriented in the true vertical plane. This data was acquired prior to machinery realignment.

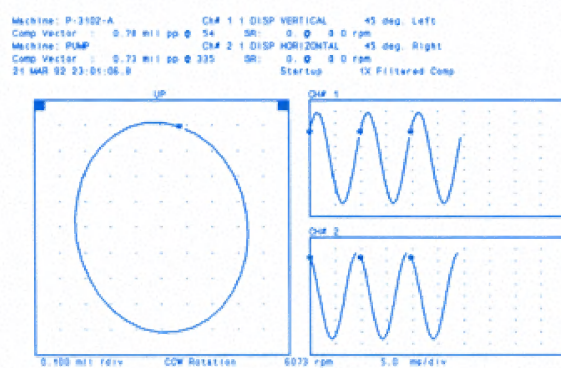


Figure 16

Shows an Orbit after realignment. Notice hardly any lateral shaft load exists.

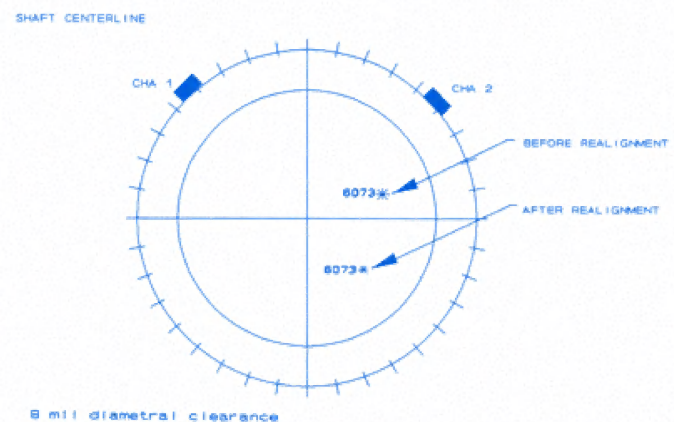


Figure 17

Before and after realignment plot of Shaft Average Centerline positions.